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# GEORGE C. MARSHALL SPACE FLIGHT CENTER,

## HUNTSVILLE, ALABAMA

; NASA TMX - 50798)

OTS: $2.60 gh, $0.95 mf

VACUUM LUBRICATION

By

K. E. Demorest

25 Jan. 1962    25p
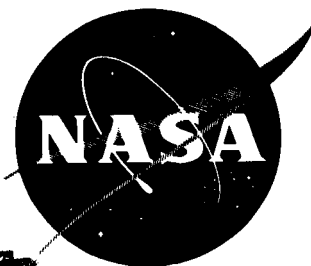
## NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

RQT-1076

GEORGE C. MARSHALL SPACE FLIGHT CENTER

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MTP-P&VE-M-62-2

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ABSTRACT                               /2 4/00

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describes the apparatus used in the study, the environment in which
evaluations of inorganic dry film lubricants are being made, and a
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                                        AUTHOR

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SUMMARY

The problem of lubrication of guidance, control, and instrument-type bearings in space is under comprehensive study. This report describes the apparatus used in the study, the environment in which evaluations of inorganic dry film lubricants are being made, and a mathematical model designed to describe the failure mode observed in actual testing. The method of total problem solution is described, and the status of current work is discussed in detail. It is noted that, although the problem is not solved, solution can be expected in a reasonable period of time.

INTRODUCTION

In the past year, the problem of lubrication in space has received considerable attention in both governmental and industrial research laboratories. Since simulation of the space environment in its totality currently is not only impractical but impossible, most investigators have selected the pressure environment of space as being the major contributor to space lubrication problems. A quick look at the pressure environment in which the bearings must operate indicates that most organic substances can be ruled out as lubricants, because of their high vapor pressures and their instability under hard radiation. Even high vacuum greases cannot be depended on to last for periods of time in excess of a few months.

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One proposed solution to vacuum, or space, lubrication problems is to hermetically seal all components requiring lubrication, thus overcoming the deficiencies of organics with respect to their vapor pressures. However, pressurized systems requiring bulky reservoirs and/or recirculation equipment for long life periods, in orders of months or years, are susceptible to failure through pressure loss and offer no means of protection of the lubricant from nuclear and/or cosmic radiation. Therefore, radiation shielding must be included in such designs, and shielding criteria cannot be established for radiation from solar flares since the magnitude of such radiation has not been determined to date. Even if such data were available, the mass of such a pressurized, radiation-shielded system would be far in excess of that obtainable by other solutions.

The most desirable solution to the problem of vacuum lubrication involves the use of dry film lubricants. Although considerable information is available on dry film lubrication and the kinetics of this process [2], [3], [4], & [5], the film materials most commonly used are organics, which are highly deteriorated by hard vacuum and/or nuclear radiation. On the other hand, inorganic dry film lubricants have been demonstrated to function satisfactorily in hard vacuum for periods of time in excess of 500 hours by a few investigators [1], and it can be established theoretically that the life of the lubricant is, among other things, a function of the film material, the bearing load, and the bearing type. Active investigation of inorganic dry film lubricants has been under experimental study by this branch for approximately two years.

Launch vehicle lubrication problems can be divided into two separate and distinctly different categories. These are, first, the lubrication of heavily loaded bearings and gears for short-time periods of an intermittent nature, such as are associated with restartable propulsion systems; and second, the lubrication of lightly loaded gears and bearings for both extended and intermittent operation times such as are found in guidance, control, and instrumentation systems. The magnitude of each of these problems, and the scope of the investigations necessary to surmount these problems are such that no small group can efficiently undertake each concurrently. Since the greatest demand of this Center is for lubricants to satisfy the latter problem, lubrication of guidance, control, and instrumentation systems, it is this area particularly which is being studied in our program, and it is to the solution of this problem that all references in this report are directed.

Current guidance, control, and instrumentation systems utilize ball or roller bearings since the coefficient of rolling or rolling-sliding friction of most materials is significantly lower than that of pure sliding friction. However, since pure sliding friction may be required for particular application, and since a solution to the lub-

rication of rolling components, in all probability, will not be appropriate for components in sliding friction, the program is designed to utilize journal bearings which operate in pure sliding friction. It is reasonable to assume that a solution to sliding friction will be equally applicable to problems involving rolling or rolling-sliding friction.

## EVALUATION PROCEDURE

The experimental program designed for the evaluation of potential inorganic dry film lubricants consists of the following criteria.

### TEST ENVIRONMENT

All testing of lubricants shall be done at a pressure of not more than $5 \times 10^{-7}$ mm Hg. Although it is recognized that pressures of several lower orders of magnitude exist in space, it can be shown theoretically that below $5 \times 10^{-7}$ mm Hg. most deterioration of inorganic materials is not significant insofar as the lubrication properties are concerned. In addition, limitations of current vacuum systems make it economically infeasible to reduce the pressure of the test chamber much below the limit established. Pressures in the $10^{-8}$ mm of Hg. range can be achieved with existing equipment when sufficient time is available to allow these pressures to be reached.

Since current data on early satellite temperature histories demonstrate that this parameter can be controlled within a reasonably narrow temperature range ($50^{\circ}C$), the ambient temperature at which testing is done shall be limited to $25^{\circ}C \pm 100^{\circ}C$.

Other environmental parameters such as nuclear, cosmic, and solar radiation are not planned currently since considerable studies on the radiation damage to materials in vacuum are being done separately under contract, and these results will be used to determine the necessity of including radiation environments in the test chamber.

The test equipment is shown schematically in FIGS. 1 and 2 and pictorially in FIG. 3. The vacuum system (FIG. 1) consists of six test bottles (1) * connected to a primary vacuum system, which is composed of a mechanical roughing pump (2) with a capacity of 13CFM, a six inch oil diffusion pump (3), and a 24 hour liquid nitrogen cold trap (4). Each test chamber can be isolated from the primary vacuum system by a vacuum valve (5) and can be pumped individually by a 75 l/min. ion pump (6). In operation, the test chamber is evacuated to a pressure in the range of $10^{-5}$ mm Hg. At this time the vacuum valve is closed, isolating the test chamber, and the ion pump is started. Upon termination of the test, the chamber is vented to the atmosphere by a vent

* Refers to location on the figure.

valve (7), and a new test is installed. After installation of the chamber, the vacuum valve is opened and the evacuation procedure is repeated. Thus, six tests can be run concurrently, and replacement chambers can be installed without interrupting the other tests.

The test equipment, FIG. 2, which is designed to fit into the test bottles, consists of a frame (8), which supports a pair of drive (ball) bearings (9), and a pair of test (journal) bearings (10). The journal bearing shaft (11) is connected to the drive bearing shaft (12) by a calibrated spring (13). The drive bearing shaft is pro- vided with a follower (14), which is coupled magnetically to a drive magnet (15). The drive magnet is located outside of the test bottle and is connected to a variable speed A. C. motor.

The measuring instrumentation for each tester is designed to permit the coefficient of friction of the test bearings to be deter- mined throughout the bearing life without interrupting the test. To do this, two square wave generators, consisting of externally mounted light sources, beam interrupters mounted to the main shaft on each side of the calibrated spring and photoelectric transducers to receive the light beam, are included in the test system. The square wave, generated by the light source on the drive bearing side of the calibrated spring, triggers an electronic counter. The wave developed by the generator on the test bearing side of the spring stops the counter. By carefully aligning the two square waves generated prior to the initiation of motion of the tester, the lag angle between the two square waves generated during motion can be translated into torque experienced by the calibrated spring through the equation

$$T = 360 \, \sigma \, \omega \, t$$

where
$T$ = torque in gm-cm
$360°$ = the number of degrees/rev
$\sigma$ = spring constant in gm-cm/degree
$\omega$ = rotational speed in rev/sec
$t$ = time lag in seconds

From this data, then, the coefficient of friction of the two journal bearings can be calculated from the equation,

$$C_f = \frac{T}{Wr}$$

where
$C_f$ = torque in gm-cm
$W$ = weight imposed on the bearings which is the weight of the test journals, shaft, and interrupters
$r$ = radius of the journal

All instrumentation for the system is automatic.  Each test is
monitored individually at preselected time intervals.  In addition to
measuring the lag angle, the rotational speed is measured by elec-
tronically isolating one of the two square wave generators and counting
the number of square waves generated by the remaining generator per
second.  All readings are made with a digital recorder, and a print-
out of all readings is made also.  Thus, the entire test operation can
proceed unattended to termination.

### TEST RESULTS

Most of the tests run to date have been made using steel journals
coated with metallic or ceramic films.  In some cases, an auxiliary
lubricant, $MoS_2$, has been used also.  After a number of runs, it became
apparent that an unsuspected mode of failure was occurring.  Table 1
illustrates the randomized type of failure occurring in some of these
tests.  It can be seen from these data that the unexplained failures
must have been caused by conditions other than lubricant wear.  In no
test was actual seizure experienced, and in all cases the bearing could
be re-started and re-run for increasingly shorter periods of time to
failure.  One condition noted in all tests, which could be used to de-
fine the failure mode, was vibration.

Due to the small clearance necessary for journal bearing operation,
it was difficult to determine the magnitude of the vibrational forces
developed.  Therefore, two journal bearings were prepared which consisted
of a full journal but only half, or 180°, bushings.  These bearings were
run to observe the magnitude of the vibration.  The journals were ob-
served to bounce as much as ¼" to ½" at certain speeds.  The magnitude
of the bouncing was not found to be directly proportional to the speed
but was noted as being variable around certain critical speeds.

Several complete test bearings were started in atmospheres of air,
helium, and nitrogen.  These bearings were allowed to run at various
speeds for periods of from a few minutes to several hours.  Subsequent
to this time, the atmosphere was removed by evacuation.  In 9 out of
10 cases, the bearings vibrated badly and stopped within 60 seconds
after removal of the gas.  It was theorized that the gas provided some
type of hydrodynamic lubrication and also acted as a vibration damper.

In order to measure the amplitude and frequency of the vibration,
a probe as shown in FIG. 4 was developed.  This consisted of a length
of spring steel, resting on the journal shaft, and attached to the
main frame.  A strain gage was mounted on the probe so that any vertical
movement of the shaft transferred a signal through an amplifier to a
Brush Electronic Recorder.  FIG. 5 and 6 are examples of actual
recordings taken with this probe.  These tests were run both in a vacuum
and in helium with ultimate failure occurring at approximately 1400 RPM

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in vacuum. From these and other tests, it has been noted that a wave envelope, FIG. 7, is set up at certain speeds, and that a sharp, stick-slip reaction seems to take place in a vacuum. The recordings in FIG. 7 show the shaft movement of a set of gold bearings coated with molybdenum disulphide running against a set of stainless steel bushings. The total run-out was .0012 inch, and rotational velocity was 2200 RPM. The last line of this recording shows the beginning of vibrational failure.

Tests have been run to attempt to correlate the effect of load and speed on vibration. Statistical analysis of these data indicate that there is little correlation between load, speed, and vibration. However, prior observations tend to indicate that some relationship does exist at least between speed and vibration. Further tests are planned to check the relationship between bearing vibration and the following variables:

1. Load

2. Speed

3. Coefficient of Friction

4. Journal Misalignment

5. Surface Hardness

Prior testing indicates that speed, coefficient of friction, and possibly surface hardness are related to bearing vibration. The capability of some bearings to function satisfactorily for a period of time before failure could be explained by the change in frictional characteristics of the lubricant caused by removal of oxides either by wear or evaporation in vacuum.

MATHEMATICAL ANALOG

It appeared that if enough of the important variables were known, it would be possible to develop a mathematical model of the reaction of the journal to these variables. The following simulation, being unidirectional and one dimensional, does not fully describe the complex motion of the journal bearing. However, it does describe one method by which a bearing could start to vibrate and fail because of vibration.

The mathematical model described in FIG. 8 shows a journal "J" on a surface "S" with a center of rotation "C" at a finite distance "b" from the geometric center "C". The radius of the journal is in-

dicated by "A". It can be seen now that the journal will have a cam effect with the distance "X" varying from (a + b) to (a - b) in 180° of rotation. This can be described by the expression,

$$\text{Sin } \sigma = \frac{b \text{ Sin } \theta}{a}$$

$$\alpha = 180° - \sigma - \theta$$

$$X_n = \frac{a \text{ Sin} \alpha}{\text{Sin } \theta}$$

Now the change in X for each increment of rotation is:

$$S_n = \Delta X = X_{n-1} - X_n$$

At the same time, the journal tends to fall toward the surface under an axial force such as gravity with a velocity $V_g$ where

$$V_g = 385.32t + V_g - 1$$

Where t = 1/6 RPM = time for one increment of revolution, 1° or 1/360 Rev.

The distance moved by the journal "J" due to gravity is

$$S_v = \frac{1}{2} V_g t$$

and the total vertical movement of the journal is

$$Sm = Sn - Sv.$$

When the journal strikes the surface of the bushing, it is assumed to rebound with a velocity

$$V_s = \sqrt{V_b^2 + V_f^2} - 385.32t$$

where the radical is a vector sum of the bounce velocity, ($V_b$), and the velocity imparted by the rotation of the journal, ($V_f$).

Then the position of the journal at any time is given by,

$$Pm = Pm-1 + Sm \quad \text{LIM } 0$$
$$\longrightarrow$$

when Pm is the present position and Pm-1 is the journal position one increment of time earlier.

The above equations were programed on a digital computer. Plots of the simulated reaction of the journal are shown in Fig. VII and VIII.

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It is assumed that failure will occur when the amplitude of the bouncing journal equals the available clearance in the bushing so that the journal strikes the opposite side. At this time, the linear acceleration will increase greatly, causing the frictional force to become very high.

FIGS. 9 and 10 show five rotations of a journal bearing at 3000 RPM, with rotation of the journal assumed to have begun at one side of the bearing. These rotations are plotted on a log scale with an assumed base line. An examination of these plots shows that on each succeeding rotation the journal rebounded higher in the bushing. Only five rotations were plotted since it was obvious that the simulated bearing was approaching failure. Computer results have simulated remarkably well the actual, physical results with regard to frequency and amplitude of vibrations, for various values of the coefficient of friction and rotational speed. Time has not permitted sufficient computer runs to determine the exact part each variable plays in the simulated vibration. However, such studies are under way. Furthermore, it is planned to expand this program into a three dimensional simulation and to determine the acceleration and friction forces involved at each increment of rotation.

Concurrent with the vibration studies, sliding friction tests are to be run using various materials on plane surfaces to eliminate rotational problems. The first tester for these experiments is completed.

CONCLUSIONS

The studies completed to date demonstrate that inorganic film lubrication of journals for vacuum operation cannot be studied in a simple manner. The kinetics and/or mechanism of vacuum lubrication with dry films bears little relationship to the hydrodynamic mechanism or atmospheric boundary layer lubrication. Until a definition of all parameters involved in vacuum lubrication is established, interim solutions to the problem will be based on a trial and error development. However, the work of this program to date demonstrates that complete definition of the problem can be made within a short time frame if approached in an organized and methodical manner. It has been determined already that vibration is a serious condition in vacuum lubrication of journal bearings. However, the vibration has been shown to be directly dependent on the following parameters:

a. Initiating axial force (e.g. gravity or centrifugal force)

b. Coefficient of friction between rotating surfaces

c. Mechanical clearance between rotating surfaces

d. Elasticity of journal and bearing films and substrates

e. Rotational velocity

f. Alignment of true centers and centroid.

Such definitions are necessary for problem solution. When the requirements are clearly defined, it is believed that suitable materials can be developed to satisfy each requirement. It is noteworthy that although ball bearings do not experience the same magnitude of vibrational forces, they do experience vibration, and the solution applicable to journal bearings will be equally applicable to ball and roller bearings.

## LUBRICATION TEST RESULTS

| BEARING MAT'L | ADD. LUBE | SPEED RPM | ENVIRONMENT | MISALIGNMENT | HOURS RUN | MODE OF FAILURE |
|---|---|---|---|---|---|---|
| Gold on Steel | None | 1000 | Vacuum | .0001 | 1/6 | Vibrated Badly and Stopped |
| Gold on Steel | $MoS_2$ | 1400 | Vacuum | .0001 | 1/6 | Vibrated Badly and Stopped |
| Gold on Steel | $MoS_2$ | 3000 | Vacuum | .0007 | 32 | Vibration Started by Striking Outside of Test Unit |
| Gold on Steel | $MoS_2$ | 5000 | Vacuum | .0010 | 632 | Unit Vibrated at Speeds Below 5000 RPM |
| Gold on Steel | $MoS_2$ | 3000 | Helium | .0001 | 232 | Unit Still Running |
| Silver on Steel | $MoS_2$ | 2200 | Vacuum | .0006 | 1/6 | Unit Vibrated Badly at Start |
| Silver on Steel | $MoS_2$ | 3000 | Helium | .0001 | 232 | Unit Still Running |
| $ALO_2$ | $MoS_2$ | 1400 | Vacuum | .0012 | 1/6 | Vibrated Badly At Start |
| $ALO_2$ | None | 3000 | Helium | .0012 | 6 | Failed Due to Black Sticky Deposit on Journal |

TABLE 1
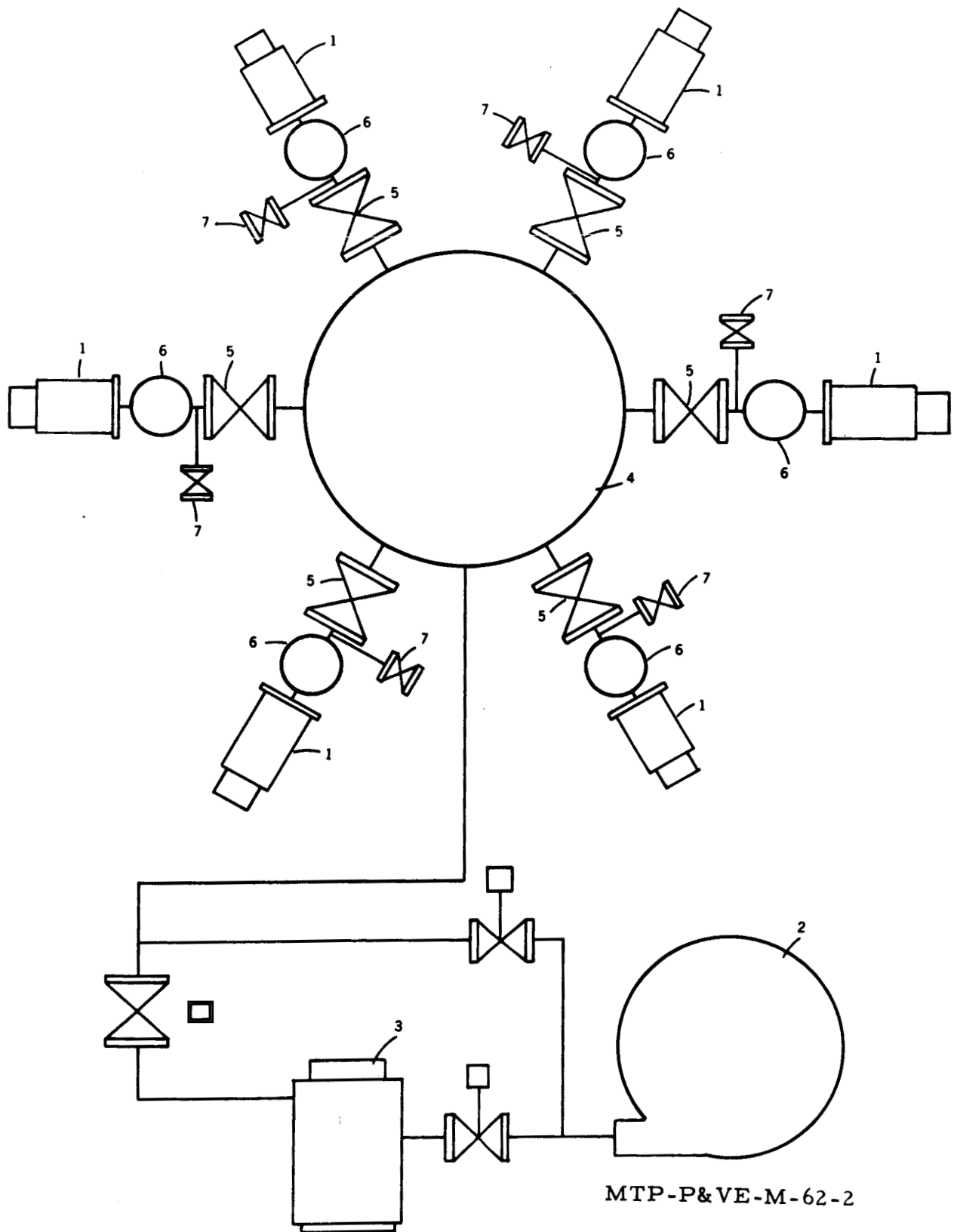
MTP-P&VE-M-62-2

Figure 1    Vacuum System

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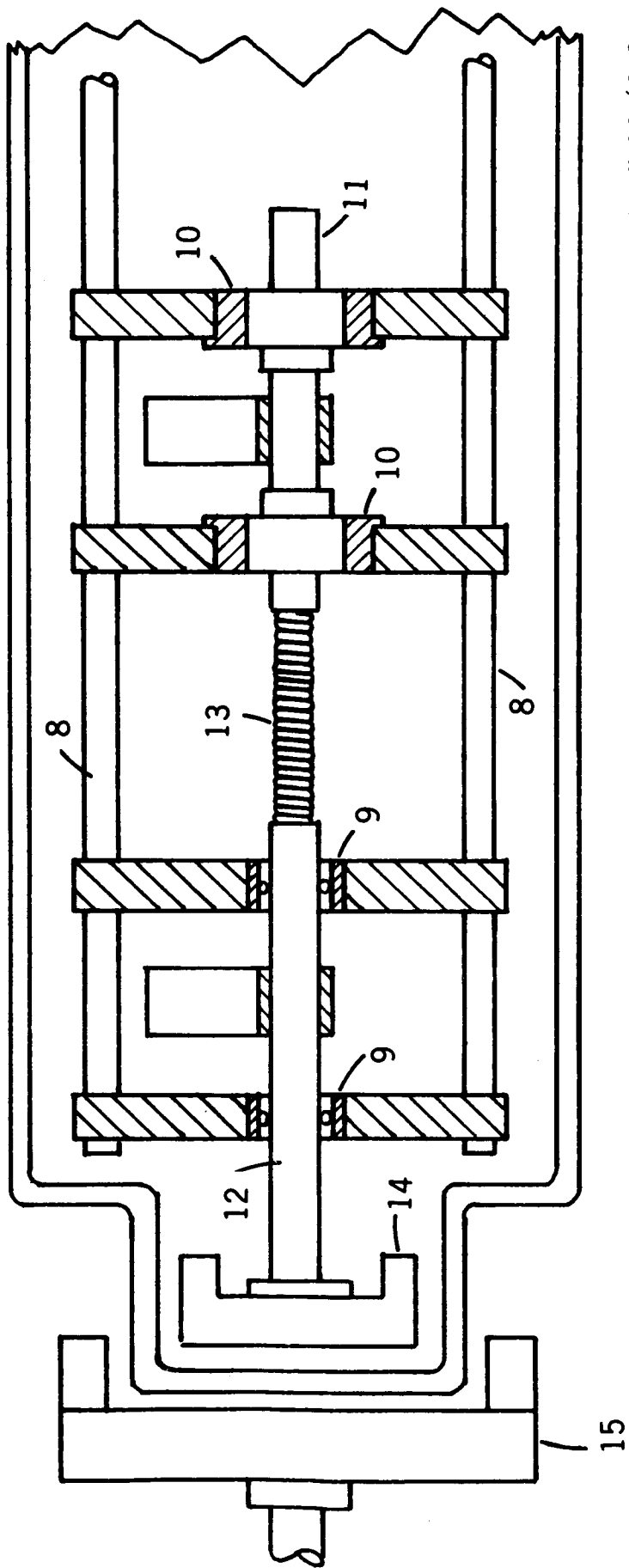
MTP-P&VE-M-62-2

Figure 2   Bearing Tester
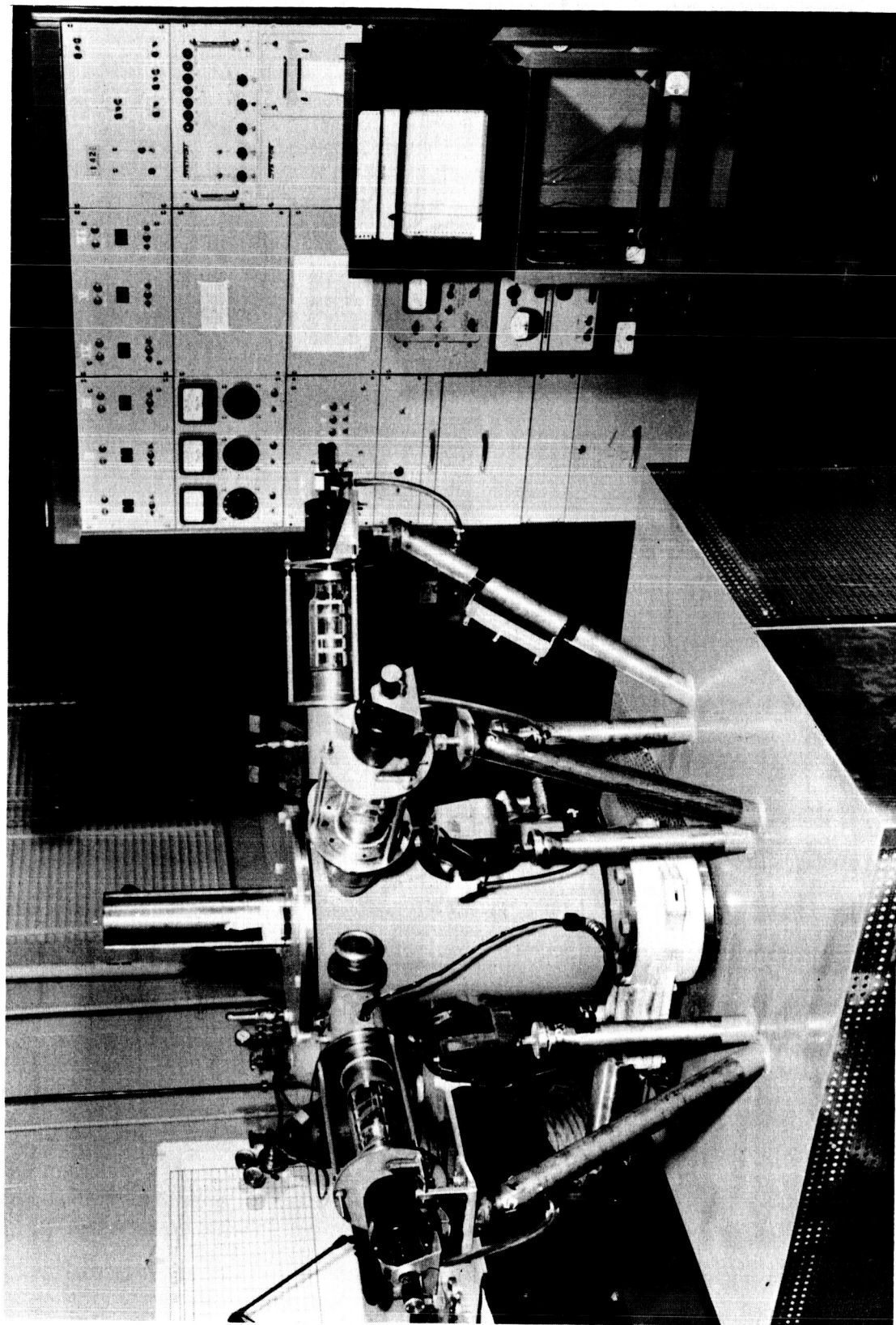
Figure 3  Vacuum Lubrication Test Equipment

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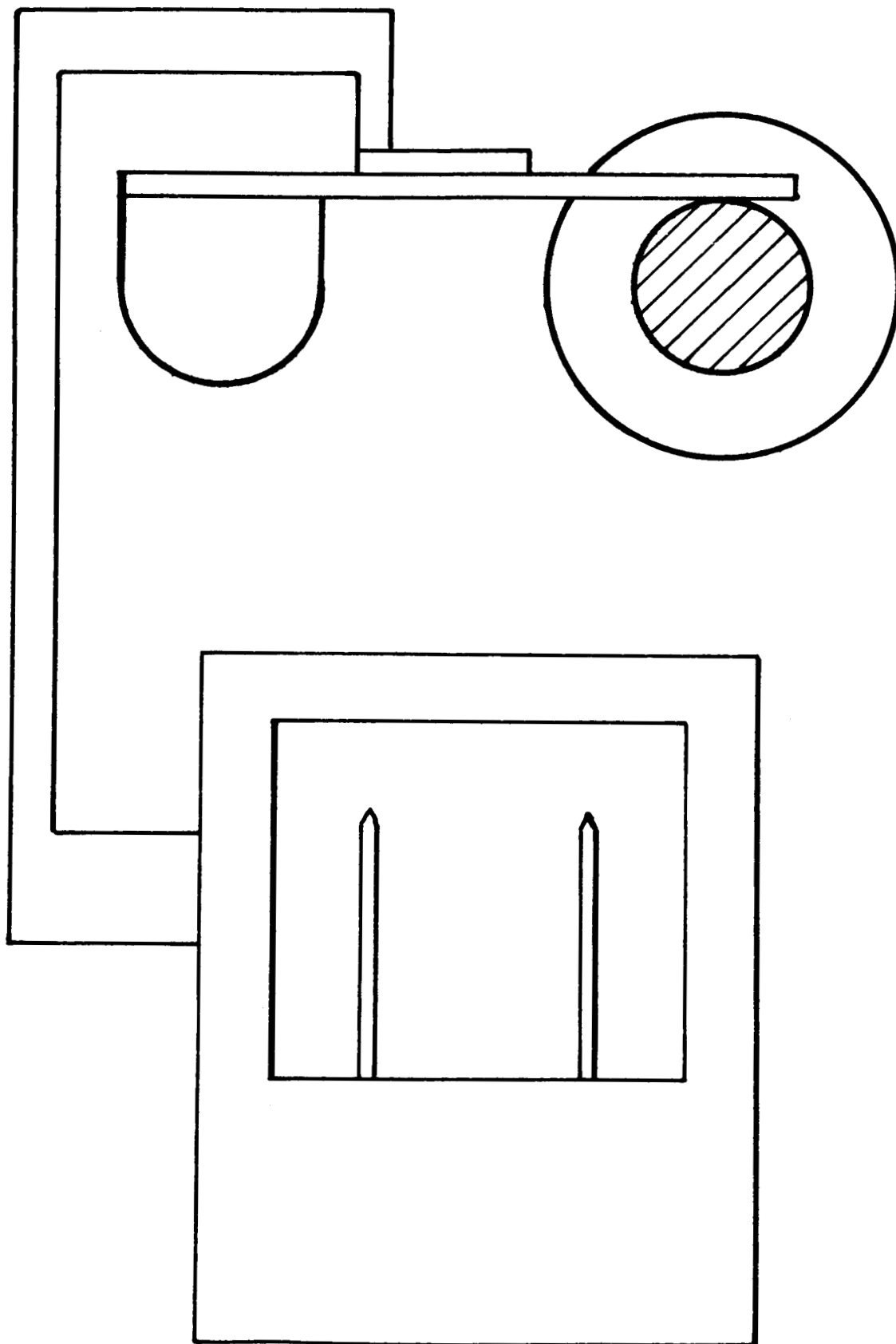


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Figure 4    Vibration Probe

1 CYCLE

600 RPM - HELIUM

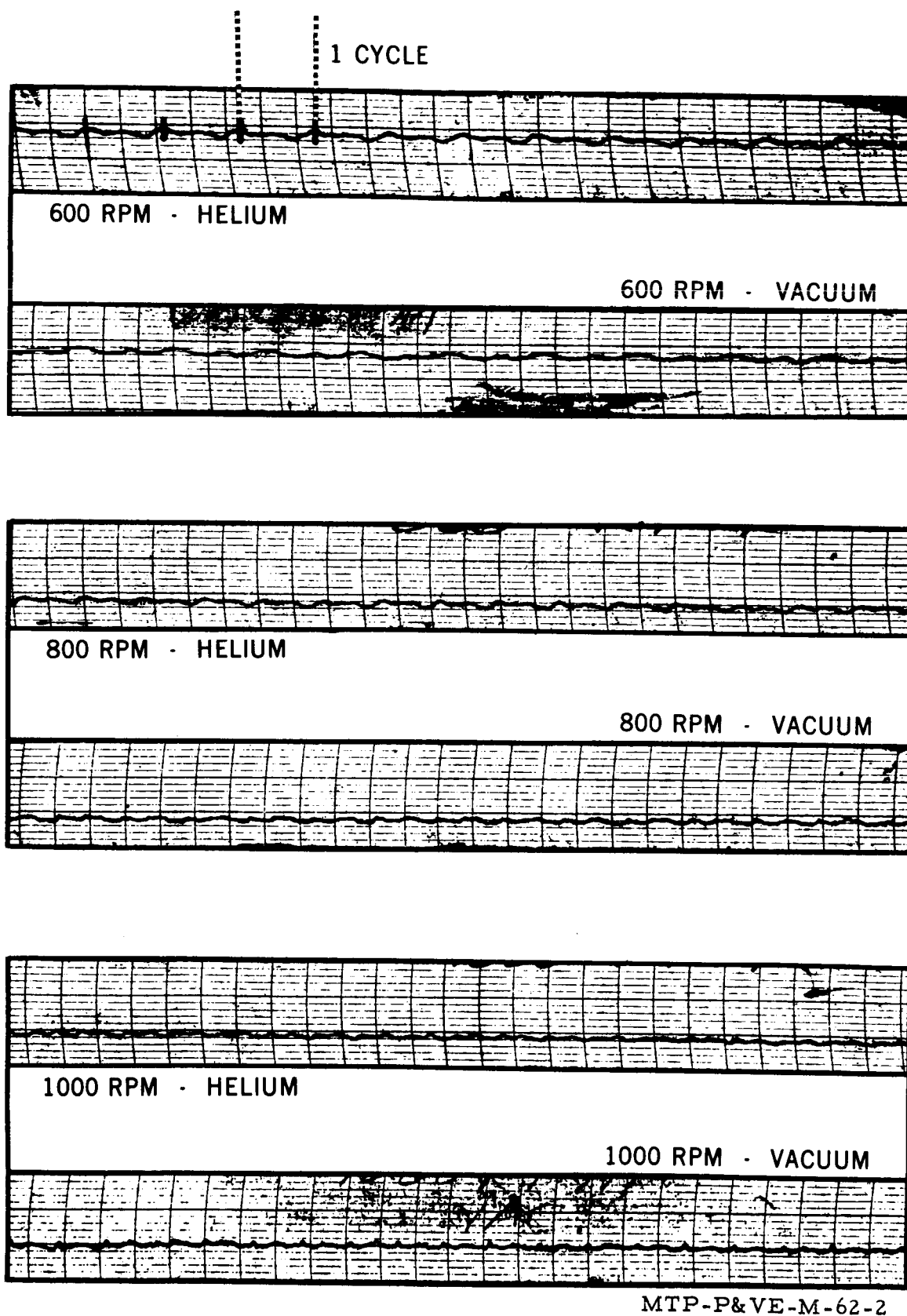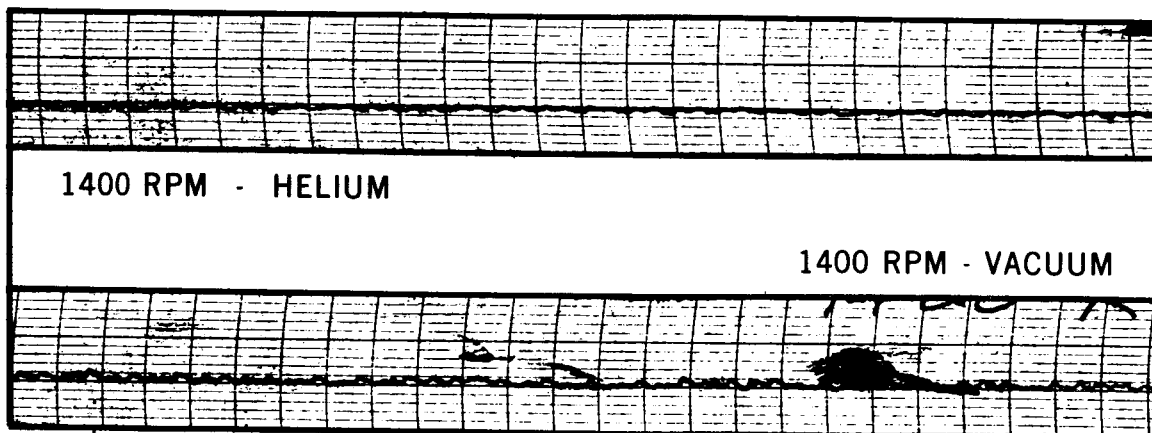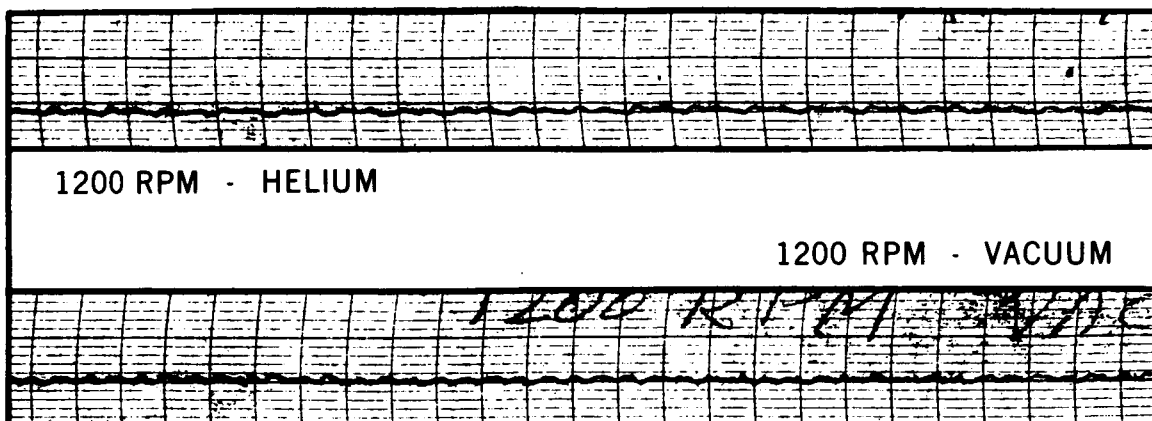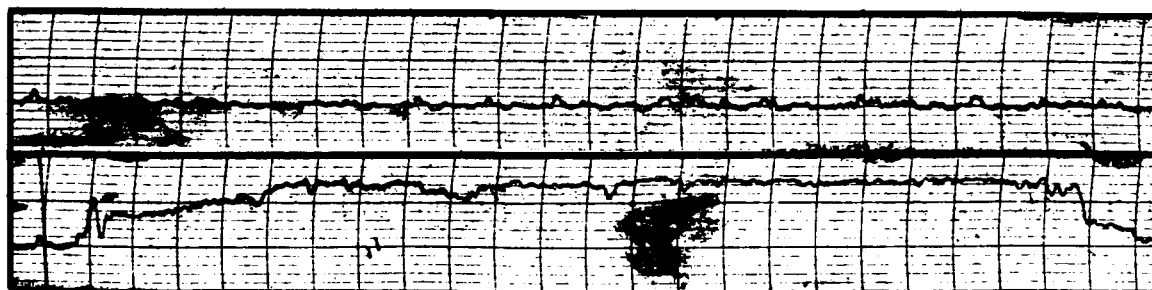600 RPM - VACUUM

800 RPM - HELIUM

800 RPM - VACUUM

1000 RPM - HELIUM

1000 RPM - VACUUM

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Figure 5    Vibration Recording

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1200 RPM - HELIUM

1200 RPM - VACUUM

1400 RPM - HELIUM

1400 RPM - VACUUM

TEST FAILED AFTER RECORDER STOPPED

STARTED AND RAN BACK UP IN VACUUM - FAILURE OCCURED BETWEEN
1000 AND 1400 RPM

MTP-P&VE-M-62-2

Figure 6    Vibration Recording
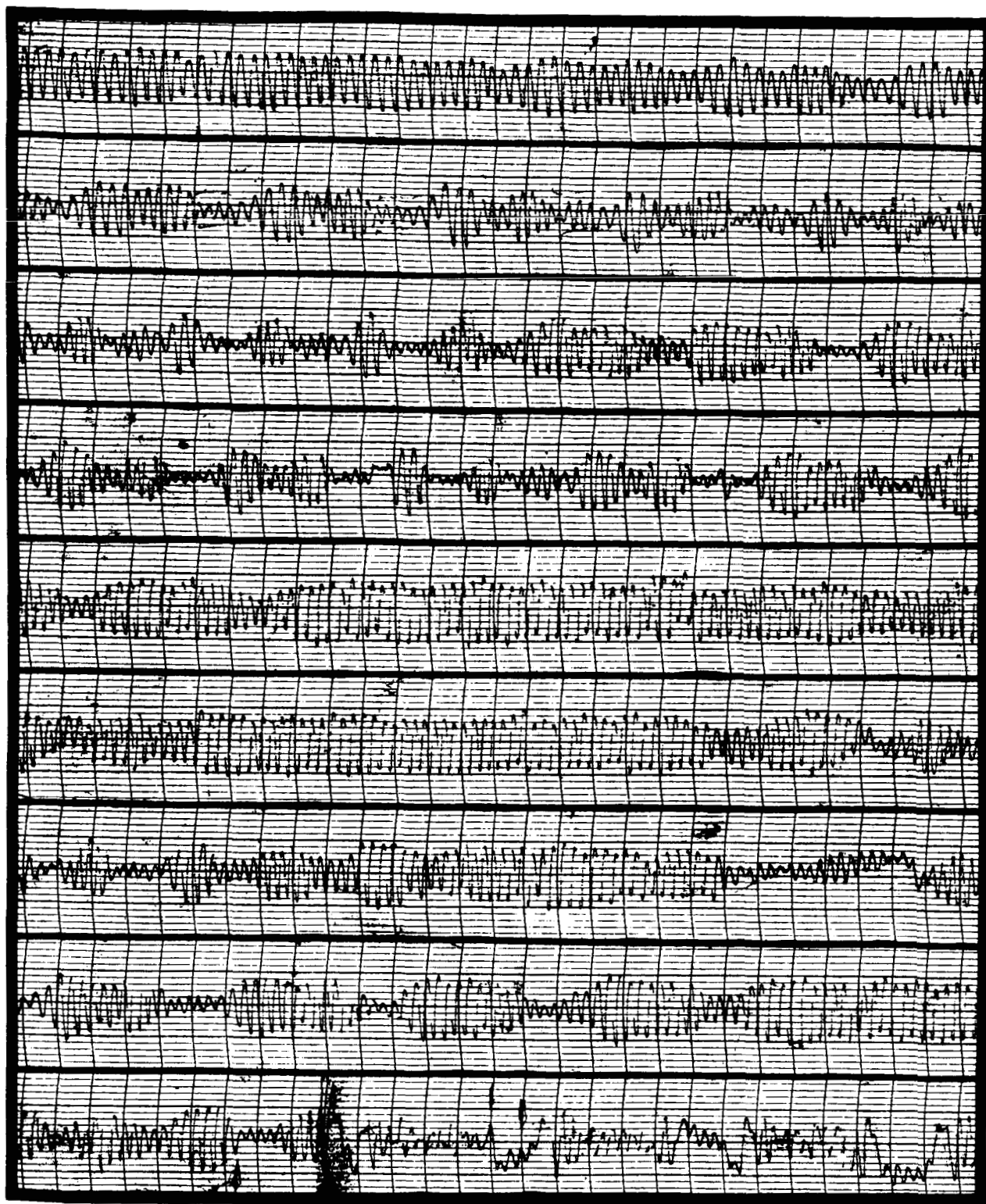
MTP-P&VE-M-62-2

Figure 7    Vibration Recording, Gold Journal on Stainless Steel
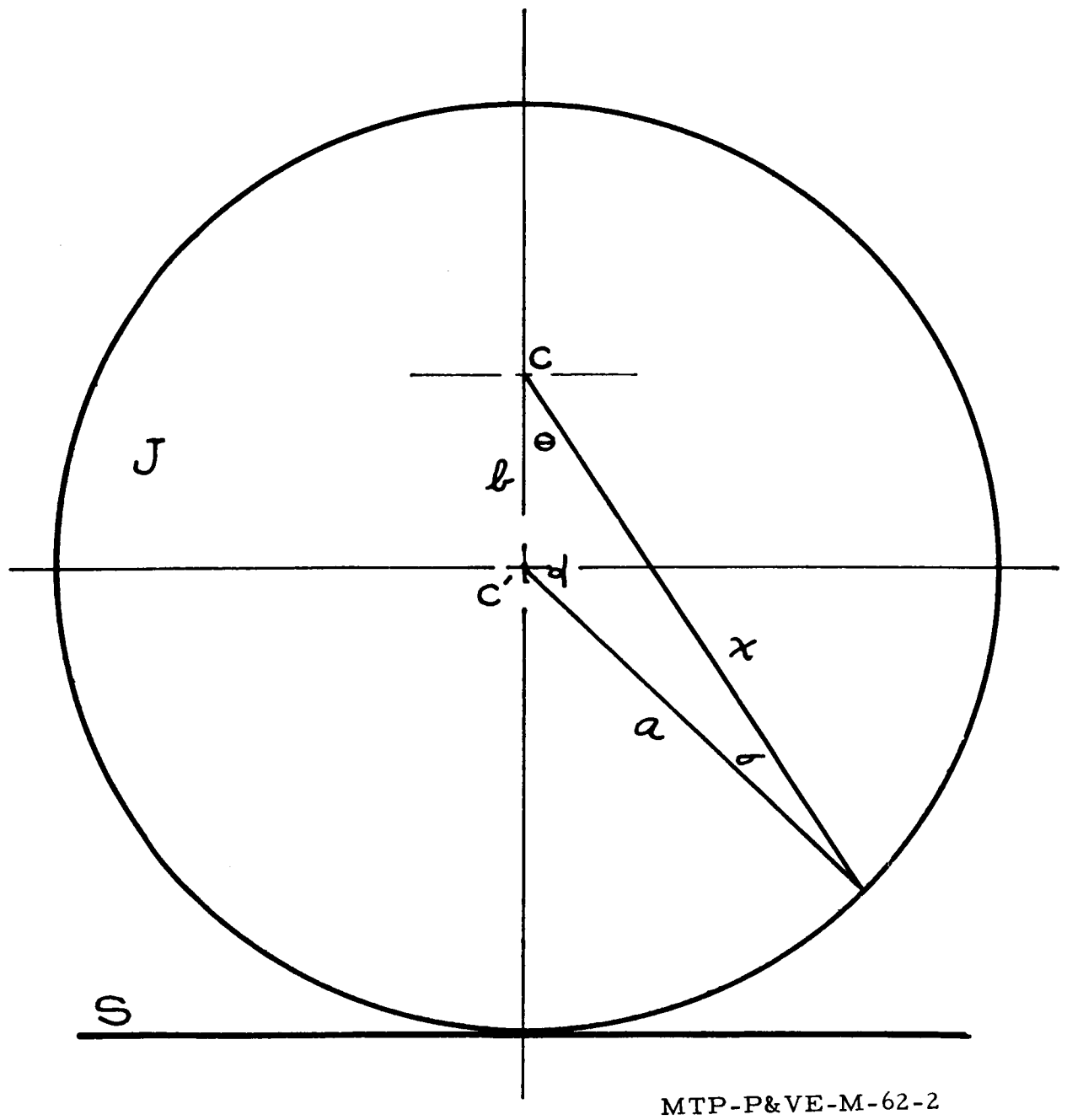Bushing with MoS₂ Lubricant  (2200 RPM)

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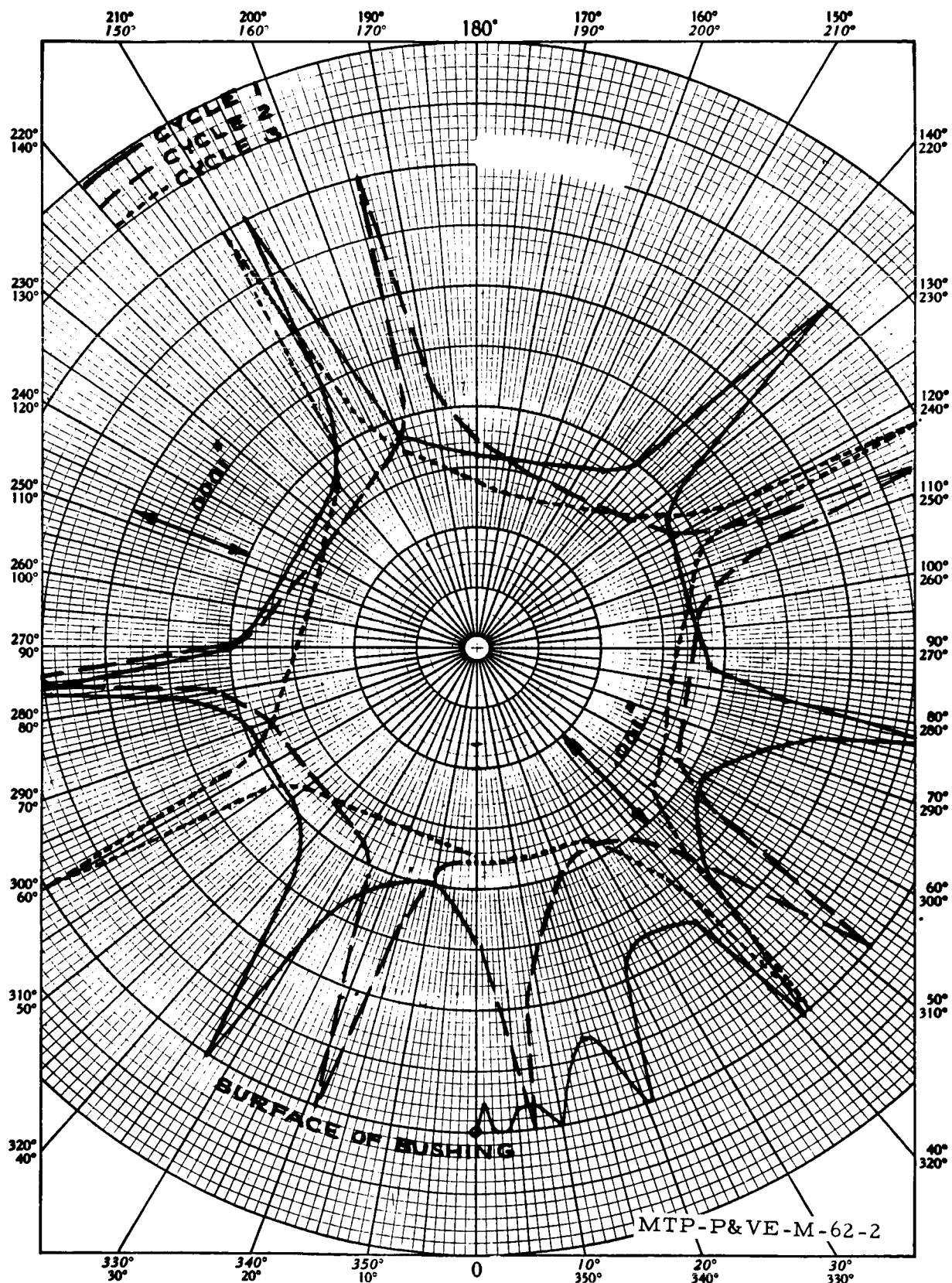Figure 8  Journal Schematic

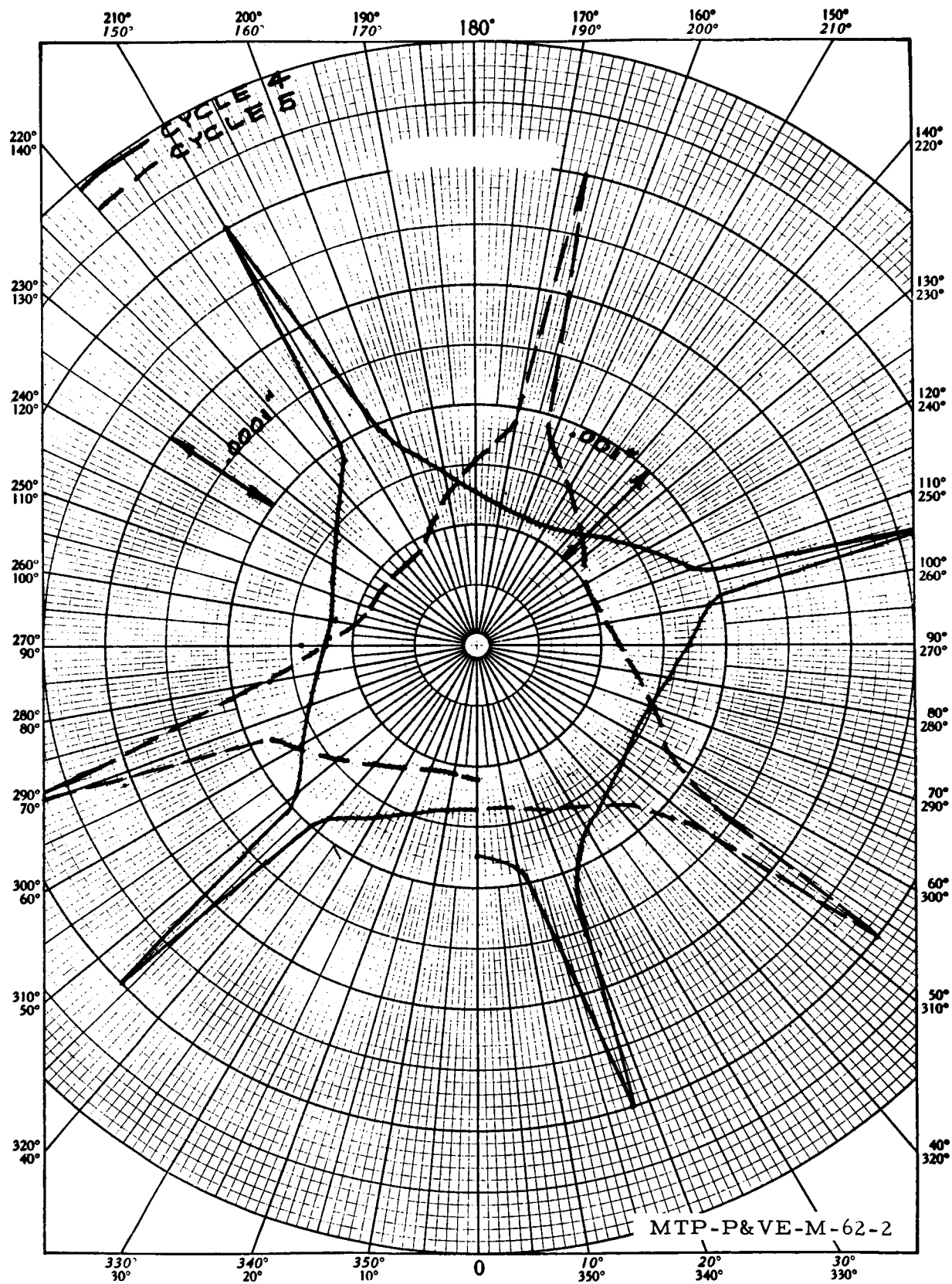Figure 9    Plot of Calculated Vibration

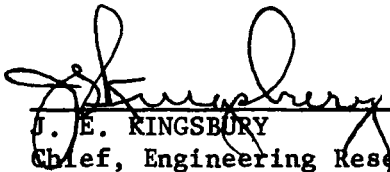Figure 10    Plot of Calculated Vibration

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MTP-P&VE-M-2-62                                    January 25, 1962

> The information in this report has been reviewed for
> security classification. Review of any information
> concerning Department of Defense or Atomic Energy
> Commission programs has been made by the MSFC Security
> Classification Officer. This report, in its entirety,
> has been determined to be unclassified.

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